DTU Library

Experimental study of the heat transfers and passive cooling potential of a ventilated plenum designed for uniform air distribution

Hviid, Christian Anker; Lessing, Julian

Published in:
CLIMA 2016 - proceedings of the 12th REHVA World Congress

Publication date:
2016

Document Version
Peer reviewed version

CLIMA 2016 - proceedings of the 12th REHVA World Congress

Heiselberg, Per Kvols

Publication date:
2016

Document Version
Publisher's PDF, also known as Version of record

Link to publication from Aalborg University

Citation for published version (APA):
Heiselberg, P. K. (Ed.) (2016). CLIMA 2016 - proceedings of the 12th REHVA World Congress: volume 5. Aalborg: Aalborg University, Department of Civil Engineering.

Experimental Study of the Heat Transfers and Passive Cooling Potential of a Ventilated Plenum Designed for Uniform Air Distribution

Christian A. Hviid[#12], Julian Lessing[*3]

[#] *Department of Civil Engineering, Technical University of Denmark*
Brovej building 118, DK-2800 Kgs. Lyngby, Denmark

[1] cah@byg.dtu.dk

[#] *Saint-Gobain Nordic*

Robert Jacobsens Vej 62A, DK-2300 Copenhagen S, Denmark

[2] christian.hviid@saint-gobain.com

[*] *Danish Energy Management & Esbensen A/S*
Galionsvej 64, DK-1437, Copenhagen K, Denmark

[3] jlj@dem-esb.dk

Abstract
Diffuse ventilation works through the pressure chamber above the acoustic suspended ceiling to uniformly distribute the supply air to the occupied zone. This, in effect, increases the thermal mass of the room because the upper slab of the room no longer is isolated from the occupied zone.
In this study, the cooling potential of a diffuse ceiling ventilation system is investigated by experiments focused toward characterizing the convective heat transfer of the plenum. The heat transfers are quantified from four different air flow rates, the temperature of the air supplied to the plenum and the mean surface temperature, i.e. the total heat transfer coefficient of the plenum.
The established heat transfer coefficient is used for analysis of the cooling performance of the system in dynamic building simulation program which showed that during peak summer days, the scenario with ventilated plenum would exhibit temperatures in the occupied zone approx. 1-1.5 °C lower than the baseline with unventilated plenum.
In conclusion this study disclosed the mean heat transfer of the plenum with an inlet jet of approx. 1.2-0.4 m/s and temperature differences of 0.5-4.5 °C and showed that ventilation supply through the plenum can be used to augment the night cooling potential.

Keywords - Diffuse ventilation, heat transfer coefficient, passive cooling

1. Introduction

The ability of diffuse ceiling ventilation (DCV) to provide draught-free ventilation in rooms with high loads has been documented [1-4], yet the benefits in terms of passive cooling are still to be examined.

The hypothesis of this study is that the DCV concept provides cooling benefits for night ventilation cooling because the upper slab is in direct contact with the ventilation air. Thus, the passive cooling potential of removing the acoustic ceiling and exposing more thermal mass in the room is significant, especially at reducing the peak temperatures [5-7].

There are two reasons why the DCV system might hold a larger cooling potential than a traditional night cooling system. One is that the heat transfer resistance between the air and the surfaces in the plenum is probably smaller than the standard value for natural convection at ceiling surfaces, due to the increased forced convection in the plenum. The hypothesis of this study is that ventilation inlet through the plenum presents a better utilization of the thermal mass of the constructions in the plenum. In literature, data for the heat transferred in and around the plenum in a DCV system in combination with night cooling is limited. The conditions may be simulated using fluid mechanics [8], or well-mixed conditions may be assumed and standard correlations used [9], but it does not take into account the effect the airflow rate, the inlet air speed or temperature differences in the plenum.

The primary goal of the experimental work in this study was to characterize the mean heat transfer of the plenum as a function of airflow rate and temperature difference and to quantify the resulting peak temperature reduction in the occupied zone [10].

2. Heat transfer methodology

The definition of the local convective heat transfer coefficient at position i, $h'_{conv,i}$ is written in (1), where Q_i is the heat flux through the surface, A_i is the defined sampling surface area, $T_{surface,i}$ is the surface temperature and $T_{air,i}$ is the local bulk air temperature.

$$h'_{conv,i} = \frac{Q_i}{A_i(T_{surface,i} - T_{air,i})} \tag{1}$$

However, building simulation tools typically regard the air temperature as fully mixed; consequently it is better to use the supply air temperature to the plenum, T_{supply} as reference instead of the local air temperature. Also the overall heat transfer coefficient accounting for the entire plenum, HTC_{plenum} is introduced, which then evaluates:

$$\begin{gathered} Q_{total} = \sum h'_{conv,i} A_i \left(T_{surface,i} - T_{supply}\right) = \\ HTC_{plenum} A_{total} \left(\bar{T}_{surfaces,average} - T_{supply}\right) \end{gathered} \tag{2}$$

Here, A_{total} is the total plenum surface area and $\bar{T}_{surface,average}$ is the area-weighted temperature of the plenum surfaces:

$$\bar{T}_{surface,average} = \frac{1}{A_{total}} \sum A_{i} T_{surface,i} \tag{3}$$

Rearranging (2) HTC_{plenum} can be expressed as:

$$HTC_{plenum} = \frac{Q_{total}}{A_{total}\left(\bar{T}_{surfaces,average} - T_{supply}\right)} = \frac{\rho q_v c_p (T_{supply} - T_{outlet})}{A_{total}\left(T_{supply} - \bar{T}_{surfaces,average}\right)} \tag{4}$$

where T_{outlet} is the temperature of the air as it exits the plenum through the diffuse ceiling.

3. Setup

The experimental room is modular, each module with a width of 3 meters. Three concrete beams with height and width 40x20 cm separate the modules. Fig. 1 depicts the situation.

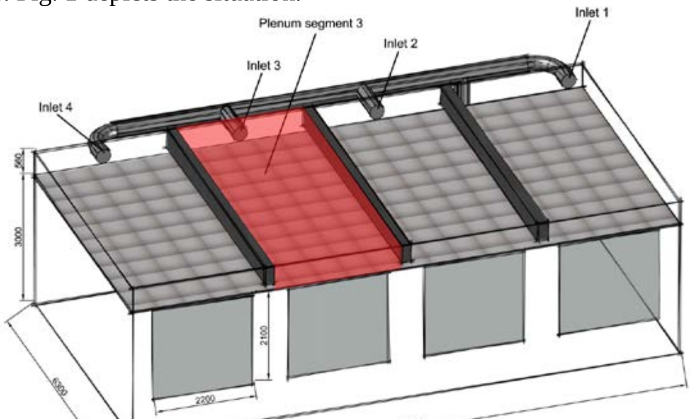

Fig. 1 Sketch of the layout of the case room with dimensions.

The plenum is partially separated into four segments by concrete beams, and for each segment there is an air inlet in a size of Ø315 mm placed on the interior wall in the plenum. The plenum segment chosen as the basis of the experimental investigation with temperature measurements is marked in red, and is referred to as “plenum segment 3”. The beams do not completely separate the air in the four plenum segments from each other; they leave a gap of 16 cm below the beams open for air to transfer between the segments.

A longitudinal section of the room is shown in Fig. 2, illustrating the layout of the plenum segmented by the beams.

The suspended ceiling in the room is made from 60x60 cm perforated gypsum tiles. On the upper side of the tiles, a thin acoustic felt open to air diffusion covers the perforations in the tiles.

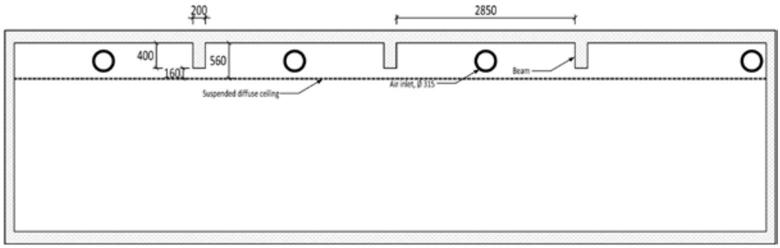

Fig. 2 Longitudinal section of the experimental room showing plenum segments and dimensions of the dividing beams.

The tiles are suspended in a reverse T-shaped suspension system and the total perforation area of the ceiling is 17 %. The entire ceiling system is identical to that described and analyzed by Hviid & Svendsen [2].

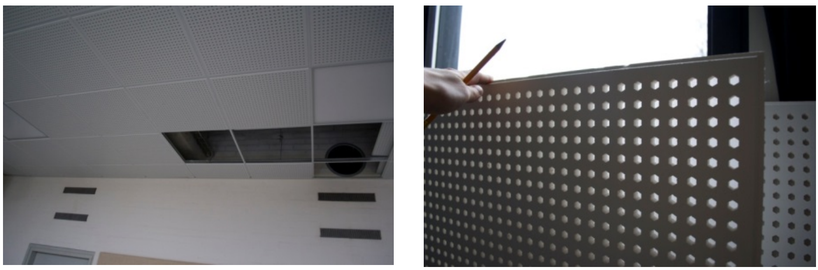

Fig. 3 Left: picture of the ceiling in the experimental room with some of the tiles moved, revealing one of the air inlets supplying the plenum with air. Right: perforated ceiling tiles

Measurements

Temperatures were measured in segment 3 only, shown in Fig. 1, 26 locations in the plenum and 8 locations in the room. This yielded a total of 34 logged temperature measurements for every time step, which was chosen to be 10 minutes. The location of the 26 measuring points in the plenum (shown in Fig. 4.) divided the plenum into 6 pieces of equal size, presuming symmetric airflow conditions around the axis of the inlet jet. Preliminary temperature measurements of bulk air and surfaces did not disclose a rotating flow pattern consistent with neither a symmetric nor a non-symmetric jet [10]. Consequently only results based on the mean temperatures are reported.

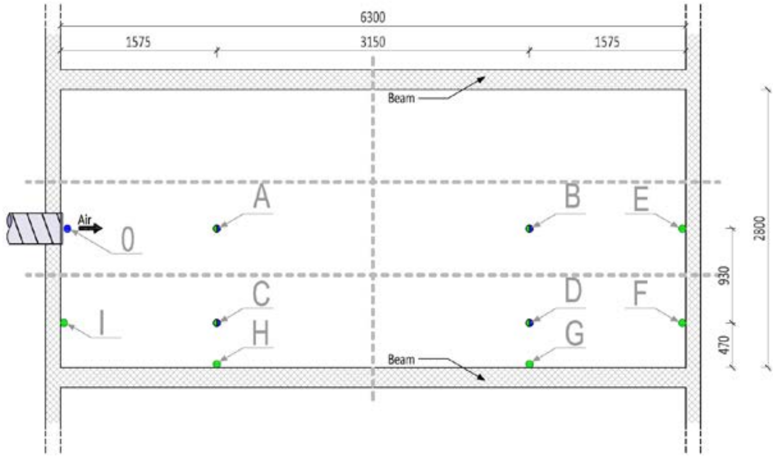

Fig. 4 Plan diagram of the measurement division in the plenum. In point A, B, C and D, two air- and three surface temperatures are measured at different heights (see Fig. 5). Air temperature are have blue markings and surface temperature have green markings.

In the center of each of these six pieces, responding to points A, B, C and D in the plan drawing, a measurement "column" of 5 points was placed as illustrated in Fig. 5.

Three of the five measurement locations were necessary to determine the heat transfer coefficient, HTC_{plenum}, i.e. the surface temperatures in point 1 and 3 according to Fig. 5, and the outlet temperature in point 5. Meanwhile, the air temperature in point 2 in the middle of the plenum is measured for a secondary examination of the temperature conditions in the plenum, and the bottom side ceiling surface temperature in point 4 is measured for an examination in relation to the thermal stratification in the room.

In Points E, F, G, H and I, the surface temperatures are measured for beams and walls, which is also needed for the determination of HTC_{plenum}. These measurements are taken in the middle of the height of the plenum, i.e. 28 cm below the concrete slab.

In point O, the temperature of the air supplied to the plenum is measured.

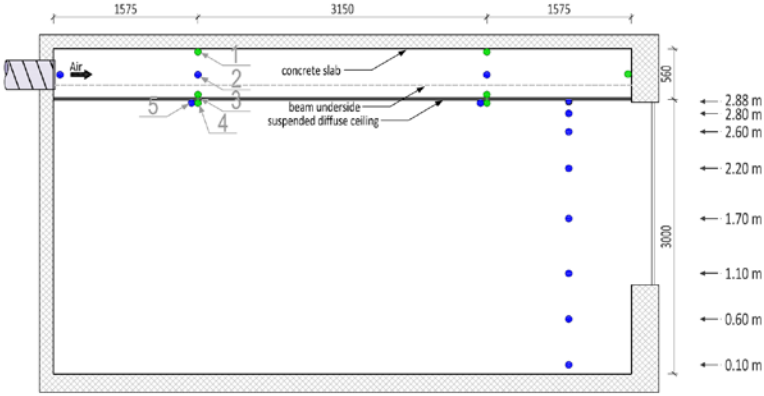

Fig. 5 Cross section of experimental. Air temperature measurements have blue marking and surface temperature green markings.

4. Results

Four different flow rates, ranging from 400-1900 m^3/h, were applied for several days each. The inlet velocity was ranging from 0.4-1.2 m/s. Typical logged values are depicted in Fig. 6. It can be seen that the average temperature of the plenum surfaces lies between room temperature and supply temperature (T_{inlet}).

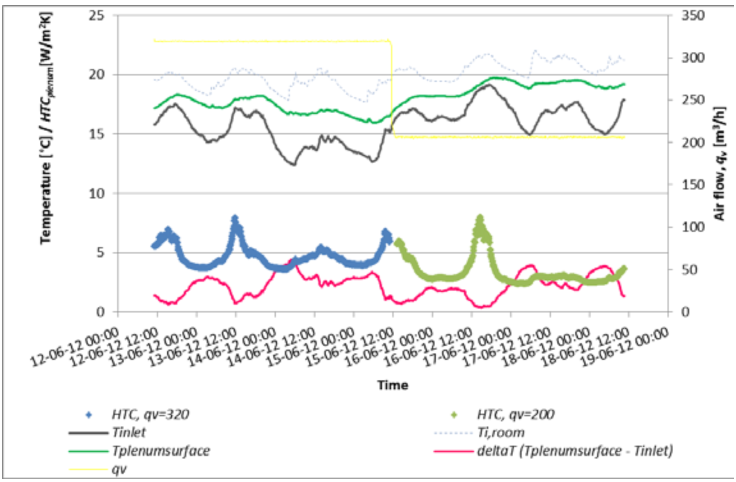

Fig. 6 Typical logged values over two periods of time with two different flow rates q_V

In Fig. 7 the temperature difference between the plenum surfaces and the supply temperature is introduced as:

$$\Delta T = \bar{T}_{surfaces,average} - T_{supply} \tag{5}$$

Is clear that the heat transfer coefficient of the plenum varies with temperature difference and flow rate, however the spread is quite large for any given temperature difference.

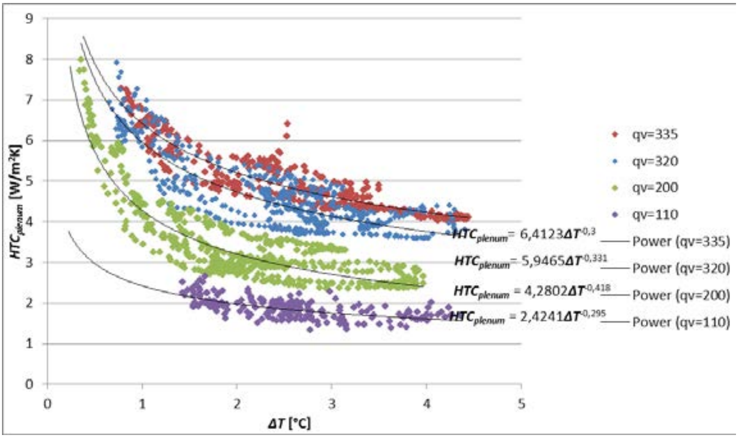

Fig. 7 Scatter plot of the heat transfer coefficient, HTC_{plenum} versus the temperature difference between supply air and mean surface temperature of the plenum surfaces, ΔT.

For each of the four applied ventilation air flows, the mean value of the heat transfer coefficient HTC_{plenum} and corresponding thermal resistance, $R_{conv,plenum}$, are given in Table 1. These values should be compared to the internal convective resistances that, for instance, are employed in the Danish commercial building simulation software Bsim [11], which are in the range 0.40-0.50 m^{2}K/W for floors and ceilings with natural convection, respectively.

Table 1. Mean values for the heat transfer coefficient, HTC_{plenum}, and the corresponding convective thermal resistance for the four different investigated air flows of the system.

q_{system} (m^3/h)	q_V (m^3/h)	q_V (l/s/m^2)	$HTC_{plenum,\ mean}$ (W/m^{2}K)	$R_{conv,plenum}$ (m^{2}K/W)
400	110	1.5	1.86	0.54
1000	200	3.7	3.42	0.29
1800	320	6.6	4.61	0.22
1900	335	7.0	5.03	0.20

Passive cooling potential

Three scenarios of a classroom were simulated; the baseline with suspended acoustic ceiling and unventilated plenum, one scenario with ventilation supply in the plenum (DCV), and for comparison, one scenario with no acoustic ceiling. The upper and lower slab and the corridor wall were all made from concrete/bricks. Thus, the thermal inertia was quite high. The convective resistances from Table 1 were implemented and radiative heat exchange was calculated during the simulation. More input information is available from [10].

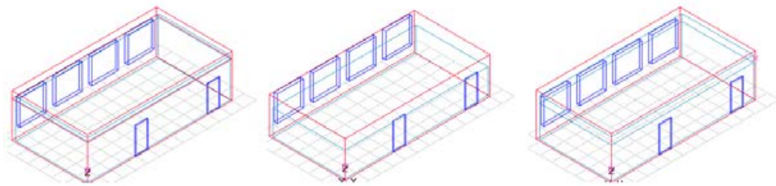

Fig. 8 Bsim models. Left: DCV, middle: baseline with unventilated plenum, right: acoustic ceiling removed

Fig. 9 shows the peak temperatures for three selected summer days. The scenario with ventilated plenum (DCV) exhibits temperatures in the occupied zone approx. 1-1.5 °C lower than the baseline and approx. 0.5 °C lower than the scenario with the acoustic ceiling removed. The cause of this is probably improved ventilative cooling of the upper concrete slab during nighttime because of the proximity between supply jet and concrete slab.

5. Conclusion

Through the experimental investigation with measurements of temperatures and air flows, the heat transfer coefficient, HTC_{plenum} was found and expressed as a function of the variables ΔT and q_v, while mean values for the heat transfer coefficient was established at the four examined air flows. In the experimental analysis, it was seen that the mean thermal convective resistance at the surfaces in the plenum can be reduced to 0.22 m^2K/W for the air flow rate of 6.61 $l/s/m^2$. For comparison, the standard value for the thermal convective resistance on ceilings in rooms where natural convection prevails, is 0.50 m^2K/W. This is a good indication of the free cooling potential, as more of the thermal mass of the constructions can be utilized as a thermal buffer.

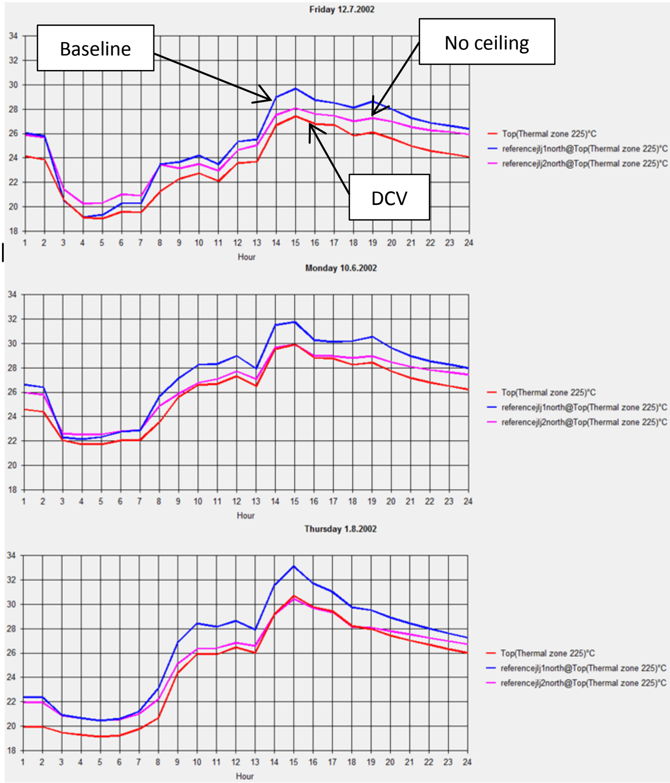

Fig. 9 Three days with peak operative temperature reductions due to improved ventilative cooling

Acknowledgment

The authors wish to express their gratitude for the financial support provided by the Bjarne Saxhof Foundation.

References

[1] P.V. Nielsen, E. Jakubowska. The performance of diffuse deiling inlet and other room air distribution systems the performance of diffuse ceiling inlet and other room air distribution systems. In: Proceedings of Cold Climate HVAC 2009, Sisimiut, Greenland, 16-19 March 2009

[2] C.A. Hviid, S. Svendsen. Experimental study of perforated suspended ceilings as diffuse ventilation air inlets. Energy Build. 56 (2013), 160–168. doi:10.1016/j.enbuild.2012.09.010

[3] J. Fan, C.A. Hviid, H. Yang, Performance analysis of a new design of office diffuse ceiling ventilation system. Energy Build. 59 (2013), 73–81. doi:10.1016/j.enbuild.2013.01.001

[4] M.H. Kristensen, J.S. Jensen. Impact of diffuse ceiling ventilation systems on indoor environmental quality in classrooms. M.Sc. thesis, Dept. of Civil Engineering, Aalborg University, 2015
[5] N. Artmann, H. Manz, P. Heiselberg. Parameter study on performance of building cooling by night-time ventilation. Renew. Energy 33 (2008), 2589–2598. doi:10.1016/j.renene.2008.02.025
[6] B. Givoni. Effectiveness of mass and night ventilation in lowering the indoor daytime temperatures. Part I: 1993 experimental periods. Energy Build. 28 (1998), 25–32. doi:10.1016/S0378-7788(97)00056-X
[7] R. Høseggen, H.M. Mathisen, S.O. Hanssen. The effect of suspended ceilings on energy performance and thermal comfort. Energy Build. 41 (2009), 234–245. doi:10.1016/j.enbuild.2008.09.006
[8] C.A. Hviid, S. Petersen. Integrated ventilation and night cooling in classrooms with diffuse ceiling ventilation, in: Proceedings of 11th Ökosan, Graz, Austria, 2011
[9] T. Yu, P. Heiselberg, B. Lei, M. Pomianowski, C. Zhang. A novel system solution for cooling and ventilation in office buildings: A review of applied technologies and a case study. Energy Build. 90 (2015), 142–155. doi:10.1016/j.enbuild.2014.12.057
[10] J.L. Jensen. Experimental and numerical analysis of the cooling potential with diffuse ceiling ventilation. M.Sc. thesis, Dept. of Civil Engineering, Tech. Univ. of Denmark, Kgs. Lyngby, 2012
[11] K.B. Wittchen, K. Johnsen, K. Grau. Bsim User’s Guide, Danish Building Research Institute, Denmark, 2011